# AIR CONDITIONER

## BACKGROUND OF THE INVENTION

The present invention relates to an air conditioner with a refrigerant circuit including a variable displacement compressor and a control valve for adjusting an opening degree in connection with variation in displacement of the compressor.

Generally, target temperature of air that has just passed through an evaporator (target after-evaporator temperature) is determined based upon cooling load information, such as ambient temperature, temperature in a compartment of a vehicle and solar irradiance. Then, the displacement of the variable displacement compressor is adjusted by a feedback control based upon the target after-evaporator temperature and actual after-evaporator temperature detected by an evaporator sensor.

A variable displacement swash plate type compressor is widely used for an on-vehicle variable displacement compressor, and a displacement control mechanism for controlling the displacement of the compressor is provided for the compressor. With respect to a control valve of the displacement control mechanism, a position of a valve body is determined by a balance between force from a pressure sensing mechanism and force from an electromagnetic actuator

so that pressure in a crank chamber is adjusted to determine the inclination angle of the swash plate, for example, as disclosed on page 8 through 11 and FIG. 3 in Unexamined Japanese Patent Publication No. 2001-173556.

Namely, the pressure sensing mechanism senses a pressure differential between two pressure monitoring points arranged in a refrigerant circuit by a pressure sensing member such as bellows and applies force based on the pressure differential to the valve body. The electromagnetic actuator strengthens and weakens the force applied to the pressure sensing member by an external control so that the set pressure differential between the two pressure monitoring points is optionally varied. The pressure differential governs internally mechanical motion of the pressure sensing mechanism. The external control of the electromagnetic actuator, that is, the variation in the set pressure differential of the control valve, is exerted based upon the target after-evaporator temperature and the detected after-evaporator temperature. In other words, when the detected after-evaporator temperature exceeds the target after-evaporator temperature, the set pressure differential is increased so that the displacement of the compressor increases. On the contrary, when the detected after-evaporator temperature is lower than the target after-evaporator temperature, the set pressure differential is reduced so that the displacement of the compressor reduces.

The pressure differential between the two monitoring points in the refrigerant circuit reflects the amount of refrigerant that flows in the refrigerant circuit. Accordingly, the amount of refrigerant that flows in the refrigerant circuit directly relates to load torque of the compressor, and the control valve directly controls the amount of refrigerant. For example, a computer for controlling a vehicle engine easily and properly estimates torque required for driving the compressor or an auxiliary machine based upon the set pressure differential (electrical signal) sent to the electromagnetic actuator of the control valve. As a result, the output of the engine is appropriately adjusted, and fuel consumption of the engine is reduced.

The electromagnetic actuator is capable of generating a small amount of electromagnetic force that can balance with a small amount of force based on the pressure differential between the two monitoring points. Accordingly, even if carbon dioxide is used as refrigerant, that is, even if pressure in the refrigerant circuit is much higher than the pressure when fluorocarbon is used as refrigerant, the enlarged electromagnetic actuator or the enlarged control valve is restrained. Namely, when the control valve of a variable set suction pressure type in which the pressure sensing mechanism operates based upon absolute value of the suction pressure requires to employ an especially large electromagnetic actuator that can generate a large amount of electromagnetic force balancing with a large amount of force based upon the suction pressure when the suction pressure

increases due to the carbon dioxide refrigerant.

An unwanted feature is that the control valve detects the pressure differential that does not reflect thermal load of the evaporator and internally and autonomically adjusts the displacement of the compressor by the feedback control. Accordingly, the set pressure differential is changed by the external control based upon the variation in the detected after-evaporator temperature due to the variation in the thermal load of the evaporator. The variation in the after-evaporator temperature slowly responds to the variation in the thermal load of the evaporator. For example, even if the thermal load of the evaporator rapidly varies, the above control valve cannot rapidly vary the displacement of the compressor. As a result, it takes a long time that the after-evaporator temperature reaches the target after-evaporator temperature so that air-conditioning feeling is deteriorated. Therefore, there is a need for an air conditioner that provides an excellent air-conditioning feeling.

## SUMMARY OF THE INVENTION

In accordance with the present invention, an air conditioner has a refrigerant circuit, a control valve, a detector, a first calculator, a suction pressure sensor and a compressor controller. The refrigerant circuit includes a variable displacement compressor. First and second pressure monitoring points are

located in the refrigerant circuit. The control valve adjusts its opening degree so as to vary a displacement of the compressor. The control valve includes an actuator and a pressure sensing mechanism that has a pressure sensing member and a valve body. The pressure sensing member autonomically detects a pressure differential between the first and second pressure monitoring points. The valve body is operatively connected to the pressure sensing member. The pressure sensing member moves in response to variation of the pressure differential, whereby the valve body is moved to vary the displacement of the compressor so as to cancel the variation of the pressure differential. The actuator changes a set pressure differential in such a manner that force applied to the valve body is changed by an external command. The set pressure differential is a reference value of a motion for determining a position of the valve body by the pressure sensing mechanism. The detector detects cooling load information in the refrigerant circuit. The calculator calculates target pressure in a relatively low pressure region in the refrigerant circuit in response to the detected cooling load information. The suction pressure sensor detects actual pressure in the relatively low pressure region in the refrigerant circuit. The compressor controller controls the actuator to eliminate a differential between the calculated target pressure and the detected actual pressure.

Alternatively, in accordance with the present invention, an air conditioner has a refrigerant circuit, a control valve, a detector, a first calculator, a surface

temperature sensor and a compressor controller. The refrigerant circuit includes a variable displacement compressor and an evaporator. First and second pressure monitoring points are located in the refrigerant circuit. The control valve adjusts its opening degree so as to vary a displacement of the compressor. The control valve includes an actuator and a pressure sensing mechanism that has a pressure sensing member and a valve body. The pressure sensing member autonomically detects a pressure differential between the first and second pressure monitoring points. The valve body is operatively connected to the pressure sensing member. The pressure sensing member moves in response to variation of the pressure differential, whereby the valve body is moved to vary the displacement of the compressor so as to cancel the variation of the pressure differential. The actuator changes a set pressure differential in such a manner that force applied to the valve body is changed by an external command. The set pressure differential is a reference value of a motion for determining a position of the valve body by the pressure sensing mechanism. The detector detects cooling load information in the refrigerant circuit. The first calculator calculates target surface temperature on the evaporator in response to the detected cooling load information. The surface temperature sensor detects actual surface temperature on the evaporator. The compressor controller controls the actuator to direct a control target to eliminate a first differential between the calculated target surface temperature and the detected actual surface temperature.

Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

## BRIEF DESCRIPTION OF THE DRAWINGS

The features of the present invention that are believed to be novel are set forth with particularity in the appended claims. The invention together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

FIG. 1 is a longitudinal cross-sectional view of a variable displacement swash plate type compressor according to a preferred embodiment of the present invention;

FIG. 2 is a longitudinal cross-sectional view of a control valve of the compressor according to the preferred embodiment of the present invention; and

FIG. 3 is a flow chart of an air-conditioning control according to the preferred embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

A preferred embodiment of the present invention will now be described with reference to FIGs. 1 through 3. The preferred embodiment applies the present invention to a vehicle air conditioner.

Now referring to FIG. 1, the diagram illustrates a longitudinal cross-sectional view of a variable displacement swash plate type compressor C according to the preferred embodiment of the present invention. A housing 11 of the compressor C defines a crank chamber or a swash plate chamber 12. A drive shaft 13 is rotatably supported by the housing 11 and extends through the crank chamber 12. The drive shaft 13 is operatively coupled to an internal combustion engine E or a drive source for traveling a vehicle through a power transmission mechanism PT.

The power transmission mechanism PT may be a clutch mechanism, such as an electromagnetic clutch, that is selective to transmit and disrupt power by an external electric control or may be a constantly transmitting clutchless mechanism, such as the combination of a belt and a pulley, that has no such clutch mechanism. Incidentally, a clutchless type power transmission mechanism is employed in the preferred embodiment.

A lug plate 14 is arranged in the crank chamber 12 and is fixedly connected to the drive shaft 13 so as to rotate integrally with. The crank chamber 12 accommodates a swash plate 15. The swash plate 15 is supported by the drive shaft 13 so as to slide and incline relative to the drive shaft 13. A hinge mechanism 16 is interposed between the lug plate 14 and the swash plate 15. Accordingly, the swash plate 15 is coupled to the lug plate 14 through the hinge mechanism 16 so that it synchronously rotates with the lug plate 14 and the drive shaft 13 and inclines relative to the drive shaft 13.

A plurality of cylinder bores 11a (only one of them shown in the drawing) is defined in the housing 11, and each of the cylinder bores 11a accommodates a single-headed piston 17 so as to reciprocate. Each of the pistons 17 engages the outer periphery of the swash plate 15 through a pair of shoes 18. Accordingly, the rotation of the swash plate 15 in accordance with the rotation of the drive shaft 13 is converted to the reciprocation of the pistons 17 through the shoes 18.

A compression chamber 20 is defined in the rear side of the cylinder bore 11a and is surrounded by the piston 17 and a valve port assembly 19 provided in the housing 11. A suction chamber 21 and a discharge chamber 22 are defined in the rear side of the housing 11.

Refrigerant gas in the suction chamber 21 is introduced into each

compression chambers 20 through a suction port 23 by pushing aside a suction valve 24 as each piston 17 moves from its top dead center to its bottom dead center. The suction ports 23 and the suction valves 24 are formed in the valve port assembly 19.The refrigerant gas introduced in the compression chamber 20 is compressed to a predetermined pressure value as the piston 17 moves from its bottom dead center to its top dead center. Then, the compressed refrigerant gas is discharged to the discharge chamber 22 through a discharge port 25 by pushing aside a discharge valve 26.

Still referring to FIG. 1, a bleed passage 27 and a supply passage 28 are provided in the housing 11. The bleed passage 27 interconnects the crank chamber 12 and the suction chamber 21. The supply passage 28 interconnects the discharge chamber 22 and the crank chamber 12. In the housing 11, a control valve CV is arranged in the supply passage 28.

The adjustment of the opening degree of the control valve CV controls a balance between the amount of discharged gas into the crank chamber 12 through the supply passage 28 and the amount of refrigerant gas out of the crank chamber 12 through the bleed passage 27 so that pressure in the crank chamber 12 is determined. A pressure differential between the crank chamber 12 and the compression chambers 20 through the pistons 17 varies in response to variation in the pressure in the crank chamber 12. Thus, the inclination angle of the swash

plate 15 is varied, and the stroke of the pistons 17, that is, the displacement of the compressor C is adjusted.

When the pressure in the crank chamber 12 is reduced, the inclination angle of the swash plate 15 increases so that the displacement of the compressor C increases. The two-dotted line of the swash plate 15 in FIG. 1 indicates a state where the lug plate 14 contacts the swash plate 15 to regulate its further inclination, that is, the swash plate 15 is at its maximum inclination angle. On the contrary, when the pressure in the crank chamber 12 is increased, the inclination angle of the swash plate 15 reduces so that the displacement of the compressor C reduces. The solid line of the swash plate 15 in FIG. 1 indicates a state where the swash plate 15 is at its minimum inclination angle.

Still referring to FIG. 1, the refrigerant circuit of the vehicle air conditioner includes the above described compressor C and an external refrigerant circuit 30. The external refrigerant circuit 30 includes a condenser 31, an expansion valve 32 and an evaporator 33.

A first pressure monitoring point P1 is located in the discharge chamber 22. A second pressure monitoring point P2 is located at a predetermined distance from the first pressure monitoring point P1 toward the side of the condenser 31 (the downstream side) in a refrigerant passage. A differential between a pressure

PdH at the first pressure monitoring point P1 and a pressure PdL at the second pressure monitoring point P2 reflects the flow rate of refrigerant in the refrigerant circuit. The first pressure monitoring point P1 communicates with the control valve CV through a first pressure introducing passage 35. The second pressure monitoring point P2 communicates with the control valve CV through a second pressure introducing passage 36 (See FIG. 2).

Now referring to FIG. 2, the diagram illustrates a longitudinal cross-sectional view of the control valve CV according to the preferred embodiment of the present invention. A valve housing 41 of the control valve CV defines a valve chamber 42, a communication passage 43 and a pressure sensing chamber 44. A rod 45 is arranged in the valve chamber 42 and the communication passage 43 so as to move in its axial direction (the vertical direction in the drawing). The upper end of the rod 45 inserted in the communication passage 43 separates the communication passage 43 from the pressure sensing chamber 44. The valve chamber 42 communicates with the discharge chamber 22 through the upstream portion of the supply passage 28. The communication passage 43 communicates with the crank chamber 12 through the downstream portion of the supply passage 28. The valve chamber 42 and the communication passage 43 constitute a portion of the supply passage 28.

A valve body portion 46 is formed at the middle portion of the rod 45 and

is located in the valve chamber 42. A step at a boundary between the valve chamber 42 and the communication passage 43 forms a valve seat 47, and the communication passage 43 serves as a kind of valve hole. As the rod 45 moves from the lowest position shown in FIG. 2 to the highest position where the valve body portion 46 seats the valve seat 47, the communication passage 43 is shut. Namely, the valve body portion 46 of the rod 45 functions as a valve body for adjusting the opening degree of the supply passage 28.

A pressure sensing mechanism includes a pressure sensing member 48 and the pressure sensing chamber 44. The pressure sensing member or a bellows spring 48 is accommodated in the pressure sensing chamber 44. The upper end of the pressure sensing member 48 is secured to the valve housing 41. The upper end of the rod 45 is fitted into the lower end of the pressure sensing member 48. The inside of the pressure sensing chamber 44 is separated into a first pressure chamber 49 and a second pressure chamber 50 by the pressure sensing member 48, which forms a cylinder with an opening at one end. The first pressure chamber 49 and the second pressure chamber 50 are respectively defined inside and outside the pressure sensing member 48. The pressure PdH at the first pressure monitoring point P1 is applied to the first pressure chamber 49 through the first pressure introducing passage 35. The pressure PdL at the second pressure monitoring point P2 is applied to the second pressure chamber 50 through the second pressure introducing passage 36.

An electromagnetic actuator 51 for changing set pressure differential is provided in the lower side of the valve housing 41. The electromagnetic actuator 51 includes a plunger housing 52 in the middle of the valve housing 41. The plunger housing 52 forms a cylinder with an opening at one end. A center post or a fixed core 53 is fixedly fitted at the opening on the upper side of the plunger housing 52. A plunger chamber 54 is defined at the lower region in the plunger housing 52 by fitting the center post 53.

A plunger or a movable core 56 is accommodated in the plunger housing 54 so as to move in its axial direction. A guide hole 57 extends through the middle of the center post 53 along its axial direction. The lower end of the rod 45 is located in the guide hole 57 so as to move in its axial direction. The lower end of the rod 45 contacts the upper end of the plunger 56 in the plunger chamber 54.

A coil spring 60 is accommodated in the plunger chamber 54 between the bottom end of the plunger housing 52 and the plunger 56. The coil spring 60 urges the plunger 56 toward the rod 45. The rod 45 is urged toward the plunger 56 by the spring property of the pressure sensing member or the bellows spring 48. Accordingly, the plunger 56 and the rod 45 regularly move upward and downward together. Incidentally, the bellows spring 48 has a greater spring force than the coil spring 60.

A coil 61 is wound outside the outer circumference of the plunger housing 52 and ranges from the center post 53 to the plunger 56. The coil 61 is supplied with electric current from a drive circuit 78 in response to a command of an air conditioner ECU or a compressor controller 72 for controlling the air conditioner. Electromagnetic force (electromagnetic attraction) corresponding to the amount of electric current supplied from the drive circuit 78 to the coil 61 is generated between the plunger 56 and the center post 53, and the electromagnetic force is transmitted to the rod 45 through the plunger 56. Incidentally, the electric current supplied to the coil 61 is controlled by adjusting applied voltage. A pulse width modulation (PWM) control is employed to adjust the applied voltage.

The opening degree of the control valve CV or the position of the valve body portion 46 of the rod 45 is determined as follows.

Still referring to FIG. 2, when no current is supplied to the coil 61 (duty ratio Dt = 0%), downward urging force of the bellows spring 48 dominantly determines the position of the rod 45. Accordingly, the rod 45 is positioned at the lowest position so that the valve body portion 46 fully opens the communication passage 43. Therefore, the pressure in the crank chamber 12 becomes maximum in accordance with the present condition, and the pressure differential between the crank chamber 12 and the compression chambers 20 through the pistons 17

becomes large. Then, the inclination angle of the swash plate 15 is minimum so that the displacement of the compressor C is minimum.

When the coil 61 is supplied with electric current that is greater than the minimum duty ratio Dt(min) in the effective range of the duty ratio (Dt(min) > 0%), the upward electromagnetic force and the urging force of the coil spring 60 exceed the downward urging force of the bellows spring 48 so that the rod 45 initiates to move upwardly. In this state, the upward electromagnetic force and the upward urging force of the coil spring 60 oppose downward pressing force based upon the pressure differential $\Delta$Pd (=PdH – PdL) and the downward urging force of the bellows spring 48. Then, the position of the valve body portion 46 of the rod 45 is determined based upon a balance among the above upward and downward urging forces. Thus, the displacement of the compressor C is adjusted.

For example, when the rotational speed of the engine E slows down to reduce the flow rate of refrigerant gas in the refrigerant circuit, the downward urging force based upon the pressure differential $\Delta$Pd weakens so that the upward urging force at the moment cannot maintain the balance between the upward and downward urging forces that act on the rod 45. Accordingly, the valve body portion 46 of the rod 45 moves upwardly to reduce the opening degree of the communication passage 43 so that the pressure in the crank chamber 12 tends to reduce. Therefore, the swash plate 15 inclines to increase its inclination

angle, and the displacement of the compressor C increases. The increased displacement increases the flow rate of refrigerant gas in the refrigerant circuit so that the pressure differential $\Delta$Pd increases.

On the contrary, when the rotational speed of the engine E speeds up to increase the flow rate of refrigerant gas in the refrigerant circuit, the downward urging force based upon the pressure differential $\Delta$Pd strengthens so that the upward electromagnetic force at the moment cannot maintain the balance between the upward and downward urging forces that act on the rod 45. Accordingly, the valve body portion 46 of the rod 45 moves downwardly to increase the opening degree of the communication passage 43 so that the pressure in the crank chamber 12 tends to increase. Therefore, the swash plate 15 inclines to reduce its inclination angle, and the displacement of the compressor C reduces. The reduced displacement reduces the flow rate of refrigerant gas in the refrigerant circuit so that the pressure differential $\Delta$Pd reduces.

Furthermore, when the duty ratio Dt supplied to the coil 61 is increased to strengthen the upward electromagnetic force, the force based upon the pressure differential $\Delta$Pd at the moment cannot maintain the balance between the upward and downward urging forces. Therefore, the valve body portion 46 of the rod 45 moves upwardly to reduce the opening degree of the communication passage 43 so that the displacement of the compressor C increases. As a result, the flow rate

of refrigerant gas the refrigerant circuit increases, and the pressure differential $\Delta$ Pd increases.

On the contrary, when the duty ratio Dt supplied to the coil 61 is reduced to weaken the upward electromagnetic force, the force based upon the pressure differential $\Delta$Pd at the moment cannot maintain the balance between the upward and downward urging forces. Therefore, the valve body portion 46 of the rod 45 moves downwardly to increase the opening degree of the communication passage 43 so that the displacement of the compressor C reduces. As a result, the flow rate of refrigerant gas the refrigerant circuit reduces, and the pressure differential $\Delta$Pd reduces.

In summary, the control valve CV internally determines the position of the valve body portion 46 of the rod 45 in response to the variation in the pressure differential $\Delta$Pd so as to maintain the set pressure differential (a target pressure differential) of the pressure differential $\Delta$Pd determined by the duty ratio Dt to the coil 61. Additionally, the set pressure differential is externally changeable by adjusting the duty ratio Dt to the coil 61.

Incidentally, the pressure in the crank chamber 12 is applied to the plunger chamber 54 through a clearance between the guide hole 57 and the rod 45. Accordingly, the pressure in the plunger chamber 54 (the pressure in the

crank chamber 12) is applied to the rod 45 to close the valve hole. Meanwhile, the pressure PdH in the discharge chamber 22 is applied to the upper end of the valve body portion 46. Accordingly, force based upon a pressure differential between the pressure PdH in the discharge chamber 22 and the pressure in the crank chamber 12 also influences on the determination of the position of the rod 45, in addition to the force based upon the pressure differential $\Delta$Pd and the force from the electromagnetic actuator 51. Namely, with respect to the control valve CV, even if the duty ratio Dt supplied to the coil 61 does not change, when there is a differential between the pressure PdH in the discharge chamber 22 and the pressure in the crank chamber 12, the set pressure differential varies a little.

Still referring to FIG. 2, the information detector 77 includes an air conditioner switch or an A/C switch 79, a temperature setting device 80, a compartment temperature sensor 81 for detecting temperature in a vehicle compartment, an ambient temperature sensor 82 for detecting ambient temperature, a solar irradiance sensor 85, a suction pressure sensor 83 and an evaporator sensor 84.

The A/C switch 79 is an ON-OFF switch of the air conditioner. The temperature setting device 80 is a device by which a passenger sets temperature in the vehicle compartment (set temperature Tset). The compartment temperature sensor 81 is a device for detecting temperature Tr in the vehicle compartment.

The ambient temperature sensor 82 is a device for detecting ambient temperature Tam. The solar irradiance sensor 85 is a device for detecting solar irradiance Ts. The suction pressure sensor 83 is a device for detecting a pressure Ps(x) in a relatively low pressure region in the refrigerant circuit, such as a suction pressure region (for example, the suction chamber 21, the inside of a conduit near the relatively low pressure region of the external refrigerant circuit 30 and an adjacent outlet of refrigerant gas in the evaporator 33). The evaporator sensor 84 is a device for detecting temperature Te(x) of air that is just passed through the evaporator 33.

Particularly, a cooling load information detector includes the temperature setting device 80, the compartment temperature sensor 81, the ambient temperature sensor 82 and the solar irradiance sensor 85. The cooling load information detector detects cooling load information in the refrigerant circuit, such as the set temperature Tset, the compartment temperature Tr, the ambient temperature Tam and the solar irradiance Ts.

The air conditioner ECU 72 adjusts the duty ratio Dt of the control valve CV, that is, the set pressure differential of the control valve CV, in response to the information detected by the information detector 77. Incidentally, The air conditioner ECU 72 not only controls the control valve CV but also, for example, controls air quantity by a conventional manner for adjusting the rotational speed

of a blower motor (not shown) in response to the information detected by the information detector 77.

Now referring to FIG. 3, the diagram illustrates a flow chart of an air-conditioning control according to the preferred embodiment of the present invention. When the engine E is started, the air conditioner ECU 72 exerts various initialization in accordance with an initial program at a step 101 (S101). For example, the air conditioner ECU 72 sets "0" as an initial value to the duty ratio Dt of the control valve CV (Namely, no electric current is supplied to the coil 61). The ON/OFF state of the A/C switch 79 is observed until it is turned on at S102. When the A/C switch 79 is turned on, the air conditioner ECU adjusts the duty ratio Dt of the control valve CV to the minimum duty ratio Dt(min) at S103 so as to start up the internally mechanical control function of the control valve CV (a function for maintaining the set pressure differential).

Required blowing temperature Ta0 of the air conditioner is calculated at S104 based upon the cooling load information (Tset, Tr, Tam and Ts) that is sent from the temperature setting device 80, the compartment temperature sensor 81, the ambient temperature sensor 82 and the solar irradiance sensor 85. The air conditioner ECU 72 serves as a calculator for calculating the target after-evaporator temperature at S105 and calculates the target after-evaporator temperature Te(set) from the calculated required blowing temperature Ta0 with

reference to map data that are previously memorized. The air conditioner ECU 72 compares the calculated target after-evaporator temperature Te(set) with the after-evaporator temperature Te(x) detected by the evaporator sensor 84 and judges whether or not a differential between Te(set) and Te(x) is equal to or less than a predetermined value (for example, 2 degrees centigrade) at S106.

When the judgment of S106 is false, that is, when the differential between Te(set) and Te(x) exceeds the predetermined value, the air conditioner ECU 72 revises the duty ratio Dt of the control valve CV so as to change the target value to the suction pressure Ps(x) detected by the suction pressure sensor 83.

Namely, the air conditioner ECU 72 serves as a calculator for calculating a target suction pressure at S107 and calculates a target suction pressure Ps(set) from the target after-evaporator temperature Te(set) calculated at S105 with reference to map data that are previously memorized. The air conditioner ECU 72 judges whether or not the suction pressure Ps(x) detected by the suction pressure sensor 83 is greater than the calculated target suction pressure Ps(set) at S108. When the judgment of S108 is false, the air conditioner ECU 72 judges whether or not the detected suction pressure Ps(x) is smaller than the target suction pressure Ps(set). When the judgment of S109 is also false, the detected suction pressure Ps(x) is equal to the target suction pressure Ps(set).

Thereby, even if the air conditioner ECU 72 does not change the duty ratio Dt of the control valve CV, it soon judges the differential between the target after-evaporator temperature Te(set) and the detected after-evaporator temperature Te(x) becomes within the predetermined value and switches a process to S116 without sending a command to change the duty ratio Dt to the drive circuit 78. Namely, as the duty ratio Dt of the control valve CV is changed, the suction pressure Ps(x) varies at first, and then the after-evaporator temperature Te(x) varies at a certain interval from the variation of the suction pressure Ps(x).

The air conditioner ECU 72 judges whether or not the A/C switch 79 is turned off at S116. When the judgment of S116 is false, the air conditioner ECU 72 switches a process to S104. On the contrary, when the judgment of S116 is true, the air conditioner ECU 72 switches a process to S101 so that the control valve CV is in a non-energized state. Thus, the displacement of the compressor C becomes minimum.

When the judgment of S108 is true, the thermal load on the evaporator 33 is regarded to be relatively large so that the air conditioner ECU 72 increases the duty ratio Dt by the unit quantity of $\Delta D$ at S110 and commands the drive circuit 78 to change the duty ratio Dt to a revised duty ratio (Dt + $\Delta D$). Accordingly, the opening degree of the control valve CV reduces a little so that the displacement of

the compressor C increases. Then, the heat removal performance rises at the evaporator 33, and not only the suction pressure Ps(x) but also the after-evaporator temperature Te(x) tends to reduce.

When the judgment of S109 is true, the thermal load on the evaporator 33 is regarded to be relatively small so that the air conditioner ECU 72 reduces the duty ratio Dt by the unit quantity of $\Delta D$ at S111 and commands the drive circuit 78 to change the duty ratio Dt to a revised duty ratio (Dt - $\Delta D$). Accordingly, the opening degree of the control valve CV increases a little so that the displacement of the compressor C reduces. Then, the heat removal performance falls at the evaporator 33, and not only the suction pressure Ps(x) but also the after-evaporator temperature Te(x) tends to increase. Additionally, the air conditioner ECU 72 switches S110 and S111 to S116.

As described above, S110 and/or S111 directs a control target to eliminate the differential between the detected suction pressure Ps(x) and the target suction pressure Ps(set). Even if the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) largely exceeds the predetermined value (for example, 2 degrees centigrade), the differential is rapidly lessened in such a manner that the duty ratio Dt is revised at S110 and/or S111. Accordingly, as coupled with the internally mechanical adjustment of the opening degree of the control valve CV, the differential between

the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) rapidly fits within the predetermined value.

When the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) is within the predetermined value by a process for revising the duty ratio Dt at S110 and/or S111, the judgment of S106 is true. When the judgment of S106 is true, a process for revising the duty ratio Dt of the control valve CV is directed to eliminate the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set).

Namely, the air conditioner ECU 72 judges whether or not the after-evaporator temperature Te(x) detected by the evaporator sensor 84 is greater than the calculated target after-evaporator temperature Te(set) at S112. When the judgment of S112 is false, the air conditioner ECU 72 judges whether or not the detected after-evaporator temperature Te(x) is smaller than the target after-evaporator temperature Te(set) at S113. When the judgment of S113 is also false, the detected after-evaporator temperature Te(x) is equal to the target after-evaporator temperature Te(set) so that the duty ratio Dt need not be changed for varying cooling performance. Therefore, the air conditioner ECU 72 switches a process to S116 without sending a command for changing the duty ratio Dt to the drive circuit 78.

When the judgment of S112 is true, the thermal load on the evaporator 33 is regarded to be relatively large so that the air conditioner ECU 72 increases the duty ratio Dt by the unit quantity of $\Delta$D at S114 and commands the drive circuit 78 to change the duty ratio Dt to a revised duty ratio (Dt + $\Delta$D). Accordingly, the opening degree of the control valve CV reduces a little so that the displacement of the compressor C increases. Then, the heat removal performance rises at the evaporator 33, and the after-evaporator temperature Te(x) tends to reduce.

When the judgment of S113 is true, the thermal load on the evaporator 33 is regarded to be relatively small so that the air conditioner ECU 72 reduces the duty ratio Dt by the unit quantity of $\Delta$D at S115 and commands the drive circuit 78 to change the duty ratio Dt to a revised duty ratio (Dt - $\Delta$D). Accordingly, the opening degree of the control valve CV increases a little so that the displacement of the compressor C reduces. Then, the heat removal performance falls at the evaporator 33, and the after-evaporator temperature Te(x) tends to increase. Additionally, the air conditioner ECU 72 switches S114 and S115 to S116.

As described above, S114 and/or S115 directs a control target to eliminate the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set). Even if the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator

temperature Te(set) exceeds the predetermined value, the differential is gradually optimized in such a manner that the duty ratio Dt is revised at S114 and/or S115. Accordingly, as coupled with the internally mechanical adjustment of the opening degree of the control valve CV, the detected after-evaporator temperature Te(x) converges in high accuracy around the target after-evaporator temperature Te(set).

According to the preferred embodiment, the following advantageous effects are obtained.

(1) The air conditioner ECU 72 revises the duty ratio Dt of the control valve CV so as to direct a control target to eliminate the differential between the detected suction pressure Ps(x) and the target suction pressure Ps(set). The suction pressure Ps(x) is physical quantity that responds to the variation of the thermal load on the evaporator 33 more rapidly than the after-evaporator temperature Te(x). Accordingly, for example, the displacement of the compressor C is rapidly varied in response to the rapid variation of the thermal load on the evaporator 33 due to the rapid variation of the rotational speed of the blower motor (air quantity). As a result, the after-evaporator temperature Te(x) rapidly approaches the target after-evaporator temperature Te(set) so that air conditioning feeling becomes satisfactory.

(2) The control valve CV is configured to subtly vary the set pressure differential when the differential between the pressure PdH in the discharge chamber 22 and the pressure in the crank chamber 12 differs, even if the duty ratio Dt supplied to the coil 61 are the same. Accordingly, in a conventional manner, for example, even if the rotational speed of the engine E (the compressor C) rapidly varies due to rapid acceleration of a vehicle and the like, that is, even if the flow rate of refrigerant gas in the refrigerant circuit rapidly varies, an external control in response to the variation of the detected after-evaporator temperature Te(x) due to the above rapid variation changes a set pressure differential to deal with the above rapid variation. Namely, the process for revising the duty ratio Dt of the control valve CV directs a control target to eliminate the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set). The above revising process still has the same problem as the prior art mentioned in the background of the invention when the rotational speed of the engine E rapidly varies. Namely, it takes a relatively long time that the after-evaporator temperature Te(x) approaches the target after-evaporator temperature Te(set). Thereby, air conditioning feeling is deteriorated.

However, the air conditioner ECU 72 in the preferred embodiment directs a control target to eliminate the differential between the detected suction pressure Ps(x) and the target suction pressure Ps(set) and revises the duty ratio Dt of the control valve CV. The suction pressure Ps(x) is physical quantity that responds to

the variation of the rotational speed of the engine E more quickly than, for example, the after-evaporator temperature Te(x). Accordingly, the displacement of the compressor C is quickly varied in response to the rapid variation of the rotational speed of the engine E, and the after-evaporator temperature Te(x) quickly approaches the target after-evaporator temperature Te(set). As a result, even if the rotational speed of the engine E rapidly varies, air conditioning feeling is satisfactory.

(3) When the differential between the target after-evaporator temperature Te(set) and the detected after-evaporator temperature Te(x) is relatively small, the air conditioner ECU 72 directs a control target to eliminate the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) and revises the duty ratio Dt of the control valve CV. Accordingly, the detected after-evaporator temperature Te(x) converges in high accuracy around the target after-evaporator temperature Te(set) so that air conditioning feeling is further improved.

The present invention is not limited to the embodiment described above but may be modified into the following alternative embodiments.

In alternative embodiments to those of the above preferred embodiment, referring to FIG. 2, the suction pressure sensor 83 is changed to a surface

temperature sensor 86 for detecting surface temperature $T_{ST}$ (temperature of a heat exchanging fin) of the evaporator 33. Additionally, a portion of the process for revising the duty ratio Dt of the control valve CV by the air conditioner ECU 72, particularly, the several steps (S107 through S111) in the flow chart in FIG. 3 are changed to steps (S107', S108', S109', S110, and S111) as follows.

Now referring to FIG. 4, the diagram illustrates a potion of flow chart that is modified from that of FIG. 3. The air conditioner ECU 72 serves as a calculator for calculating target surface temperature $T_{ST}$(set) at S107' and calculates the target surface temperature $T_{ST}$(set) from the target after-evaporator temperature Te(set) calculated at S105 with reference to map data that are previously memorized. The air conditioner ECU 72 judges whether or not the surface temperature $T_{ST}$(x) detected by the surface temperature sensor 86 is greater than the calculated target surface temperature $T_{ST}$(set) at S108'. When the judgment of S108' is false, the air conditioner ECU 72 judges whether or not the detected surface temperature $T_{ST}$(x) is smaller than the target surface temperature $T_{ST}$(set) at S109'. When the judgment of S109' is also false, the detected surface temperature $T_{ST}$(x) is equal to the target surface temperature $T_{ST}$(set).

Thereby, the air conditioner ECU 72 soon judges the differential between the target after-evaporator temperature Te(set) and the detected after-evaporator temperature Te(x) is within the predetermined value (for example, 2 degrees

centigrade) without changing the duty ratio Dt of the control valve CV and switches a process to S116 without commanding the drive circuit 78 to change the duty ratio Dt. Namely, as the duty ratio Dt of the control valve CV is changed, the surface temperature $T_{ST}(x)$ of the evaporator 33 varies at first. Then, the after-evaporator temperature Te(x) varies at a certain interval from the variation of the surface temperature $T_{ST}(x)$.

When the judgment of S108' is true, the thermal load on the evaporator 33 is regarded to be relatively large so that the air conditioner ECU 72 increases the duty ratio Dt by the unit quantity of $\Delta D$ at S110 and commands the drive circuit 78 to change the duty ratio Dt to a revised duty ratio ($Dt + \Delta D$). Accordingly, the opening degree of the control valve CV reduces a little so that the displacement of the compressor C increases. Then, the heat removal performance rises at the evaporator 33, and the surface temperature $T_{ST}(x)$ of the evaporator 33 and the after-evaporator temperature Te(x) tend to reduce.

When the judgment of S109' is true, the thermal load on the evaporator 33 is regarded to be relatively small so that the air conditioner ECU 72 reduces the duty ratio Dt by the unit quantity of $\Delta D$ at S111 and commands the drive circuit 78 to change the duty ratio Dt to a revised duty ratio ($Dt - \Delta D$). Accordingly, the opening degree of the control valve CV increases a little so that the displacement of the compressor C reduces. Then, the heat removal performance falls at the

evaporator 33, and the surface temperature $T_{ST}(x)$ of the evaporator 33 and the after-evaporator temperature Te(x) tend to increase.

The surface temperature $T_{ST}(x)$ of the evaporator 33 is physical quantity that responds to the variation of the thermal load on the evaporator 33 more quickly than the after-evaporator temperature Te(x). Accordingly, the same advantageous effects to those mentioned in the paragraphs (1) through (3) of the preferred embodiment are obtained.

In the above preferred embodiment, when the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) is within the predetermined value, the air conditioner ECU 72 directs a control target to eliminate the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) and revises the duty ratio Dt of the control valve CV. Furthermore, when the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) exceeds the predetermined value, the air conditioner ECU 72 directs a control target to eliminate the differential between the detected suction pressure Ps(x) and the target suction pressure Ps(set) and revises the duty ratio Dt of the control valve CV. In alternative embodiments to those of the above preferred embodiment, irrespective of the differential between the target after-evaporator temperature Te(set) and the detected after-evaporator

temperature Te(x), the air conditioner ECU 72 directs a control target to eliminate the differential between the detected suction pressure Ps(x) and the target suction pressure Ps(set) and revises the duty ratio Dt of the control valve CV. Namely, for example, S106 and S112 through S115 are omitted from the flow chart in FIG. 3 in the above preferred embodiment. Even so, when the rotational speed of the engine E or the thermal load on the evaporator 33 rapidly varies, the displacement of the compressor C is quickly varied so that air conditioning feeling is satisfactory.

In alternative embodiments to those of the above preferred embodiment, referring to FIG. 5, the diagram illustrates a portion of flow chart that is modified from that of FIG. 3. The control target in connection with the process for revising the duty ratio Dt(x) of the control valve CV is changed in response to large and small of the set pressure differential of the control valve CV, that is, large and small of the duty ratio Dt(x) supplied to the coil 61. Namely, when the duty ratio Dt(x) of the control valve CV is within the predetermined value Dt(set), that is, when the flow rate of refrigerant gas in the refrigerant circuit is controlled in a relatively large flow rate range, the air conditioner ECU 72 directs a control target to eliminate the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) and revises the duty ratio Dt(x) of the control valve CV. On the contrary, when the duty ratio Dt(x) is less than the predetermined value Dt(set), that is, when the flow rate of refrigerant

gas in the refrigerant circuit is controlled in a relatively small flow rate range, the air conditioner ECU 72 directs a control target to eliminate the differential between the detected suction pressure Ps(x) and the target suction pressure Ps(set) and revises the duty ratio Dt(x) of the control valve CV.

Thereby, a control of the flow rate in the refrigerant circuit is stable in a relatively small flow rate range so that air conditioning feeling is satisfactory. Namely, the control valve CV is configured to detect the pressure differential $\Delta$ Pd between the pressure monitoring points in the refrigerant circuit and internally and autonomically exerts a feedback control of the displacement of the compressor C. Accordingly, when the flow rate of refrigerant gas in the refrigerant circuit is relatively small, the variation of the pressure differential $\Delta$ Pd in response to the variation of the flow rate of refrigerant gas is relatively small (not clear) so that the internally mechanical control of the control valve CV does not properly function. As a result, when the air conditioner ECU 72 directs a control target to eliminate the differential between the detected after-evaporator temperature Te(x) and the target after-evaporator temperature Te(set) for revising the duty ratio Dt(x) of the control valve CV, the flow rate control of the refrigerant circuit in a relatively small flow rate range becomes unstable due to a slow response of the detected after-evaporator temperature Te(x) in response to the revising of the duty ratio Dt(x).

In alternative embodiments to those of the above preferred embodiment, referring to FIG. 1, a first pressure monitoring point P1' is located at a suction pressure region between the evaporator 33 and the suction pressure chamber 21 including the evaporator 33 and the suction pressure chamber 21 in the refrigerant circuit, while a second pressure monitoring point P2' is located downstream to the first pressure monitoring point P1' in the same suction pressure region.

In alternative embodiments to those of the above preferred embodiment, the control valve CV employs a bleed side control valve that adjusts the pressure in the crank chamber 12 by adjusting the opening degree of the bleed passage 27 instead of the supply passage 28.

In alternative embodiments to those of the above preferred embodiment, the variable displacement compressor employs a wobble type.

Therefore, the present examples and embodiments are to be considered as illustrative and not restrictive, and the invention is not to be limited to the details given herein but may be modified within the scope of the appended claims.